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Chapter 7—Ventilation

Hazim B. Awbi*

The University of Reading, Dept of Constr. Management & Eng., Reading RG6 6AW, UK

1. Introduction

1.1. *Historical background*

Building ventilation is necessary for supporting life by maintaining acceptable levels of oxygen in the air, to prevent carbon dioxide (CO_2) from rising to unacceptably high concentrations and to remove odour, moisture and pollution produced internally. Although CO_2 is not considered a harmful gas, nevertheless a high CO_2 concentration (e.g. above 5000 ppm) is synonymous with deficient oxygen levels in the air.

In the past, ventilation has been applied to buildings to remove excess heat in hot climates. In the Middle East wind towers were developed during the first millennium to scoop the cool wind into the building, which was sometimes made to pass over water cisterns to produce evaporative cooling and a feeling of freshness. In temperature climates, such as in Middle and Northern Europe, ventilation was mainly used to remove smoke produced by hearths, rather than to provide fresh air for breathing [1]. In the 18th and 19th centuries ventilation of particularly small dwellings became a social problem in Europe and, consequently, scholars and medical professionals started to assess the need for providing fresh air to buildings in terms of quantity and methodology. Until Pettenkofer in Germany showed in 1862 that CO_2 is harmless, it was always perceived as the cause of 'bad air'. In England, Barker [1] advocated 1000 ppm of CO_2 as an appropriate concentration which was thought to be sufficient to render odour unnoticeable. This corresponds to a fresh air supply rate of about 7 l s^{-1} per person.

The concentration of CO_2 was considered by many as the criterion for admitting fresh air into a building. However, some have doubted the suitability of CO_2 concentration as an index of air quality. More recently, research has shown [2] that, in modern buildings, other pollutants can be more important in terms of quantities produced and their impact on human health.

Over the years, ventilation guides have revised the recommended fresh air supply

* Corresponding author. Tel.: +00-44-118-931-6786; Fax: +00-44-118-931-3856; E-mail: h.b.awbi@reading.ac.uk

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rates to building occupants. Figure 1 [3] shows the changes in recommended fresh air rates in the USA during the last 160 years. If anything, this figure shows our lack of knowledge, even today, of the optimum fresh air rate that a designer is required to provide a building with. This is partly due to design changes, technological development, changes in lifestyle and also to the relative cost of energy during any one period of time when these ventilation rates were specified.

More recently, the identification of Sick Buildings (SB) and the coining of the phrase Sick Building Syndrome (SBS) has again focused attention to the method used in ventilating a building, i.e. whether air-conditioning or natural ventilation should be used, as well as the quantities of fresh air supplied to a building. Some studies [4] have found a link between the occurrence of SBS and ventilation although other factors were also found to have an influence. It is suggested that lack of fresh air is a contributory factor but not necessarily the main cause of SBS. So we now find ourselves, as was the case throughout the last three centuries, in a position where we know that ventilation is a necessity but we are not sure about what a building, or an occupant, requires for health and comfort. We do not claim that an answer to this question will be found in this chapter, but the intention here is to discuss the problems involved and provide the means to quantify ventilation rates that meet certain usage and requirements of buildings and their occupants.

1.2. *What is ventilation?*

In the last two decades, ventilation and energy conservation have been the main themes of building system design. Ventilation has become a science among building

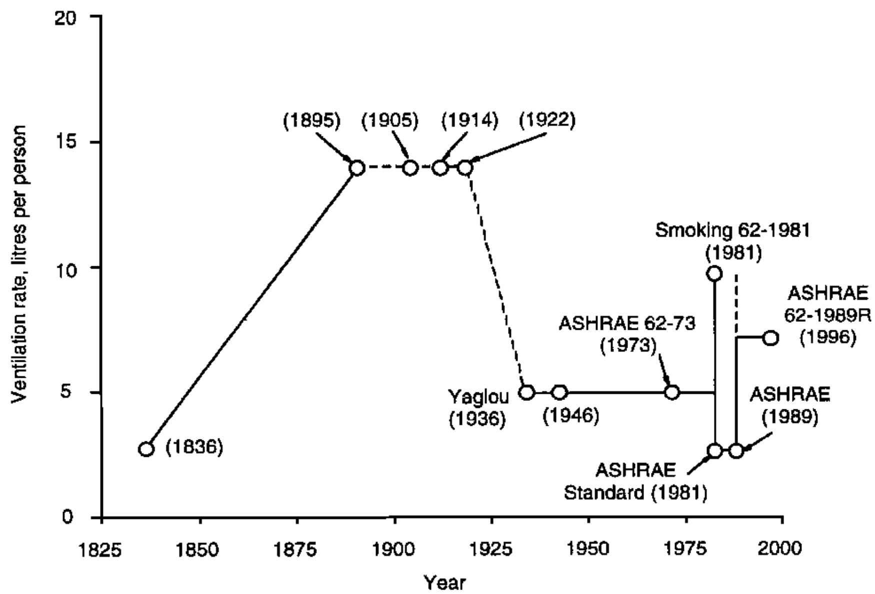


Fig. 1. Changes in the minimum ventilation rates in the USA.

system designers, scientists and engineers, who are concerned with the well being of the building occupants. Major international conferences which are dedicated to this topic, e.g. Roomvent, Indoor Air, Healthy Buildings etc., have been held periodically over the last two decades.

As a result of the increasing interest in the subject, new concepts have emerged covering wide aspects of ventilation. It is not sufficient now just to assume that ventilation air can be introduced from any convenient opening in a room at a rate taken from some design guide because there are many parameters which can influence the perceived air quality indoors. For example, the method of air distribution is so important in influencing not only the air quality in the occupied zone but also the cooling or heating energy requirement for the space. New terms such as ventilation effectiveness, ventilation efficiency, air quality index, age of air, etc., have all become important indices for assessing the ventilation process [5]. It is no longer sufficient just to specify $X\,\mathrm{l\,s^{-1}}$ of fresh air to the space, as this will have little meaning unless it is assessed in the context of these ventilation indices. So, nowadays we consider ventilation as the process of providing fresh air to the building occupants, rather than the buildings themselves, in order to sustain a good standard air quality with minimum capital cost and environmental impact. The need for ventilation is still the same as it has been over many decades, namely to provide oxygen for breathing and removing the internally produced pollution. What is changing, however, is the means of achieving this need.

1.3. *Ventilation requirements*

As mentioned earlier, ventilation is required for breathing and the removal of internally produced pollution. Before a ventilation rate can be specified, it is first necessary to estimate the rate of production of all known pollutants, viz. body bioeffluents, carbon dioxide, tobacco smoke, volatile organic compounds (VOCs), ozone, particulates, radon, water vapour, etc. Because it is not known how the aggregate of all such pollutants affect the air quality, these are normally considered separately, i.e. a ventilation rate is estimated for each known pollutant and the largest value is used for design purposes. If the air is to be heated or cooled, then it may be necessary to recirculate some of the room air but this component is excluded from the ventilation rate because it is already polluted. The pollution concentration levels that can be tolerated in buildings are found in guides and standards, such as ASHRAE Standard 62-1989R [6] and the UK Health and Safety Executive (HSE) guidelines [7]. HSE specifies two exposure limits: the long term exposure limit (LTEL) and the short term exposure limit (STEL) for dealing with different exposure periods to various pollutants.

Another factor which is often ignored but is very important in determining ventilation rates is the ventilation effectiveness (ε_v) which is defined as:

$$\varepsilon_v = (c_o - c_i)/(c - c_i) \times 100(\%) \tag{1}$$

where

c_i = pollution concentration in the supply air, ppm or mg m^{-3}
c_o = pollution concentration in the exhaust air, ppm or mg m^{-3}
c = mean pollution concentration in the occupied zone, ppm or mg m^{-3}.

The value of ε_v depends on the ventilation strategy used, i.e. location of air supply and extract openings, the momentum and turbulence of the supply air and the room heat load and its distribution. Values of ε_v can only be obtained by measurements or simulation of the air movement using computational fluid dynamics (CFD), or may be found in handbooks or guidelines for certain air distribution strategies. As an example, a typical value of ε_v for high level mixing ventilation might be around 70%, whereas for floor displacement ventilation it is somewhere in the region of 120%. Hence, theoretically at least, based on these values a displacement system should require only about 58% of the ventilation rate of a high level system. Further information on air distribution in rooms is given in Section 7 and ref [8].

2. Indoor pollutants

There are many pollutants present in room air at any one time, some of which exist at such low concentrations that they are considered harmless to the occupants, whereas others can be at high concentrations. An estimation of the main pollutants is necessary to calculate fresh air rates. It is not possible in this chapter to describe all known internal pollutants and, therefore, only some of the most common pollutants which are known to be present in room air are described.

2.1. *Odour*

Odour is a bioeffluent associated with occupancy, cooking, bathroom activities and waste. Although the experience of odour is not pleasant, it does not normally affect health. Body odour is emitted by sweat and sebaceous secretion through the skin and by the digestive system. The results of the tests which were carried out by Yaglou et al. [9] on people in a ventilated test chamber some 60 years ago are still in use today. Yaglou's work showed that the air supply rate is dependent on the occupation density of the space in m^3 per person, i.e. as the volume increases the air supply rate required to achieve an acceptable odour intensity is reduced. However, more recent research by different investigators has failed to find a correlation between room volume and the required fresh air rate.

It is generally accepted that a minimum fresh air rate of 3 l s^{-1} per person will be required to dilute body odour [6]. This is in addition to the ventilation rates required to dilute pollution from buildings, their contents and any HVAC system that may contribute to indoor pollution.

2.2. *Carbon dioxide*

The main source of CO_2 indoors is the building occupants. The rate of production of CO_2 by respiration is directly proportional to the metabolic rate through the relationship:

$$G = 4 \times 10^{-5} M A$$

where

$G = CO_2$ production rate, l s^{-1}
M = metabolic rate, W m^{-2}
A = body surface area, m^2

An average sedentary adult produces about 5 ml s^{-1} (18 h^{-1}) of CO_2 by respiration.

Most ventilation standards recommend that the CO_2 concentration should not exceed 0.5%, or 5000 ppm. However, more recent studies have indicated that this limit is far too high for human comfort. At this concentration, the occupants can experience headaches and lethargy and 1000 ppm is now widely accepted as a limit for comfort. Using eqn (2), it is possible to estimate ventilation rates based on CO_2 emission if one assumes that an average outdoor CO_2 concentration is about 400 ppm which could vary depending on whether the building is an urban, suburban, or rural location.

2.3. *Tobacco smoke*

The risk to health from tobacco smoke is widely publicized and the odour from smoke constitutes irritants to the eyes and the nasal passages, such as acrolein, tar, nicotine and carbon monoxide. British Standard 5925 [10] recommends 8, 16, 24 and 36 l s^{-1} per person of fresh air for rooms with no smoking, some smoking, heavy smoking and very heavy smoking respectively. Because of the additional flow rate required, an estimate of the smoking population should be made at the design stage. For this purpose, statistical data on the smoking population in the country should be used where available.

2.4. *Formaldehyde*

Formaldehyde (HCHO) is a chemical that is commonly used in the manufacture of building materials, furnishing, cosmetics, toiletries, food packaging, etc. The inexpensive urea-formaldehyde (UF) resin is the most commonly used adhesive in the production of plywood, wood chipboard, hardboard, plaster board and as a binder in the production of fiberglass insulation. Formaldehyde polymers are also used in the manufacture of wallpapers, carpets and textiles. Unvented combustion appliances are also a source of formaldehyde indoors and so is tobacco smoke.

Formaldehyde can enter the body through inhalations, ingestion or adsorption. It is a strong irritant and genotoxic, i.e. once in the body formaldehyde rapidly reacts with tissues containing hydrogen and damages them. Although not conclusive, there is a belief that formaldehyde poses a carcinogenic risk to humans.

The emission rate of formaldehyde in buildings depends, among other things, on the age of the material. Peak emission rates are produced by new products due to the presence of free formaldehyde molecules. For a formaldehyde source of a given age, the emission rate to room air depends on the area of emitting surface, total air volume,

Table 1
Typical formaldehyde emission rate

Material	Emission rate ($mg\ h^{-1}\ m^{-2}$)
Woodchip boards	0.46–1.69
Compressed cellulose boards (e.g. hardboard)	0.17–0.51
Plasterboards	0–0.13
Wallpaper	0–0.28
Carpets	0
Curtains	0

air change rate and other parameters, such as temperature and humidity. The steady state unsuppressed emission is expressed by:

$$C = AE/(\rho NV)$$

where

C = formaldehyde concentration, ppm
A = area of emitting surface, m^2
E = net emission rate from surface, mg m^{-2} h^{-1}
ρ = density of air, kg m^{-3}
N = air change rage, h^{-1}
V = room air volume, m^3

If the emission is suppressed, such as for very low air change rates, then the formaldehyde concentration in room air will increase and the emission rate will continuously decrease until it falls to zero at zero air change rate. Typical formaldehyde emission rates from common building materials and furnishings are given in Table 1.

Most current ventilation guides and standards recommend a maximum exposure limit to formaldehyde of about 0.1 ppm. Even though this conception has been found to be excessive for individuals who are sensitive or sensitizable, this limit has been found to be exceeded for several type of dwellings, particularly those insulated with UF foam insulation.

2.5. *Ozone*

Ozone (O_3) is naturally present in outdoor air, but its concentration is dependent on altitude and climate. It is also produced indoors by electrostatic appliances and office machines, such as photocopiers and laser printers. It has the potential for adverse acute and chronic effects on humans if present in high concentrations. The World Health Organisation (WHO) recommends a maximum concentration of 100 $\mu g\ m^{-3}$ (50 ppb) for an 8 h exposure and ASHRAE Standard 62-1989 R [6] suggests 235 $\mu g\ m^{-3}$ (120 ppb) for a short term exposure of 1 h.

Table 2
Classification of VOC exposure-effects [11]

Concentration range (μg m^3)	Description
< 200	Comfort range
200–3000	Multifactorial exposure range
3000–25,000	Discomfort range
> 25,000	Toxic range

2.6. *Volatile organic compounds* (*VOCs*)

Volatile organic compounds (VOCs) are produced indoors from a variety of sources. There is, however, no clear definition of the classes of VOCs present in indoor air, though researchers define these as compounds having boiling points between 50 and 260°C. Although formaldehyde is considered a VOC it is usually dealt with separately because it requires different measuring techniques than those used for most other VOCs. In indoor air measurements VOCs are often reported as total volatile organic compounds (TVOCs). These are usually given as the sum of the concentrations of the individual VOCs. Research on the health effects of VOC is relatively new and there is little information available on the long term exposure to most known VOCs. There are, however, some exceptions such as formaldehyde whose health effects are better understood. In most buildings the concentration of VOCs is not sufficiently large to be able to establish their health risk. Field studies in some European countries did not find a positive correlation between measured indoor air TVOC concentrations and sick buildings syndrome (SBS) prevalence. There are, therefore, no established LTEL or STEL limits for TVOC in indoor air, although Molhave [11] conducted laboratory studies of the responses of human subjects exposed to controlled concentrations of 22 VOCs mixture. As a result of these studies we may be able to classify the exposure effect of VOCs as shown in Table 2. However, the concentrations in most buildings are usually much lower than those given in the table.

2.7. *Radon*

Radon is a radioactive gas which is present in small amounts in the earth's upper crust. The gas itself is harmless but for the alpha particles emitted by short lived decay 'daughters'. These particles normally have small penetration depths and they only form a health risk if inhaled; damage to the lining of the lungs could occur posing the risk of cancer.

The concentration of radon in the atmosphere is measured in picocuries per litre (pCi l^{-1}) or bequerels per m^3 (Bq m^{-3}), where 1 pCi l^{-1} = 37 Bq m^{-3}. The concentration of radon daughters is measured in terms of the working limit (WL) which is the equivalent to alpha particle emission of 1.3×10^5 MeV per l, i.e. 1 WL = 100 pCi l^{-1}.

The concentration levels of radon in buildings depends on the geological history of the site, hence there is a wide variation in concentration levels even in one country. Unless high concentrations of radon is known to exist in the locality of a building, no special treatment is needed. In high risk areas the most effective way of reducing radon concentration indoors is by extracting air from underneath the ground floor of the building. A radon concentration value of 0.01 WL is usually used as a limit for calculating ventilation rates.

2.8. *Particulates*

Particulates suspended in air (aerosols) form a major source of indoor air pollution. Depending on their size and room air movement, aerosols can deposit on a surface within minutes or remain airborne for weeks. The constituent of aerosols can be dust, dander, fibrous, pollen, fungi, moulds, mites, bacterial, viruses, etc. Dust particles smaller than about 0.5μm can accumulate in the lung lining, causing blockages to the respiratory tubes. Biogenic particles can transmit disease or allergy. The most effective way of reducing the concentration of aerosols is by air filtration.

2.9. *Water vapour*

Water vapour exists in outdoor air and is also produced in buildings by occupants activities and certain processes. Water vapour itself does not represent a health risk. Recent research has shown that the lower the humidity is the better the perceived indoor air quality becomes. However, it is generally believed that low humidity levels contribute towards increased risk of infection of the respiratory tract and high humidity levels can cause discomfort, due to the inhibition of sweat. Furthermore, certain types of buildings require precise control of humidity to sustain its content or the processes being carried out. It is, therefore, necessary in some buildings to control the humidity of the air by means of an HVAC system.

3. Ventilation strategies

3.1. *Ventilation rates*

The ventilation rate required for a given room or a building is determined to satisfy both health and comfort criteria. The health criterion should take into consideration the exposure of the occupants to indoor pollutants which will involve the identification of the pollutants, their sources, source strengths and a knowledge of the short term exposure limit (STEL) or long term exposure limit (LTEL) for the pollutants. These limits are used to estimate the ventilation rate required to obtain the pollutant concentration that can be tolerated. Where the location of the pollution sources can be identified, the preferred approach would be for the removal of such pollutants at source.

The comfort criterion, however, will produce ventilation rates that can minimise

the effect of odour and sensory irritants from occupants' bioeffluents, occupants' activities and pollutants emitted from the building, the building systems and furnishings. This is usually used in domestic buildings, office buildings, public buildings, etc. and the health criterion is applied to industrial buildings. Despite their different chemical composition and sensory effects, studies have shown that pollutants can have additive impact called 'agonism' both in terms of smell and irritation. However, details of how agonism can be assessed are not available and as an approximation it is suggested that all source strengths (due to people and buildings) be added for the calculation of a design ventilation rate.

ASHRAE Standard 62-1989R gives two methods of determining ventilation rates: the prescriptive procedure and the analytical procedure. In the prescriptive procedure, tables of ventilation rates required to dilute the pollution produced by people and buildings are given for different types of buildings. In the analytical procedure, the ventilation rates are calculated using data for pollution sources and the effectiveness of the ventilation system. Details of these two procedures are given in ref. [6].

The European CEN pre-standard pr ENV 1752 [12] proposes the use of three categories A, B, C of buildings and recommends a ventilation rate accordingly. An air supply rate of 10 l s^{-1} per person which corresponds to 15% of the occupants predicted dissatisfied (PD) for category A; 7 l s^{-1} per person which corresponds to 20% PD for category B; and 4 l s^{-1} per person for a category C building which corresponds to 30% PD.

The expression below can be used to calculate the ventilation rate, Q, required to maintain the concentration of a particular pollutant within a desired value:

$$Q = G/\{\varepsilon_v(c_i - c_o)\} \times 10^6 \text{ m}^3\,\text{s}^{-1} \qquad (2)$$

where

G = pollutant generation rate, $m^3\,s^{-1}$ or kg s^{-1}
c_i = indoor concentration that can be tolerated, ppm or mg kg^{-1}
c_o = outdoor concentration of the pollutant, ppm or mg kg^{-1}
ε_v = effectiveness of ventilation system.

3.2. *Energy implications of ventilation*

In modern and retrofit buildings, ventilation is probably the greatest component of the total energy consumption. This is usually in the range of 30–60% of the building energy consumption. The large proportion of ventilation energy is due to three main reasons. Firstly, modern buildings are generally well insulated and, therefore, the heat gain or loss through the fabric is low. Secondly, modern building materials and furnishings emit large amounts of VOCs and TVOCs and so it becomes necessary to dilute their concentration by supplying greater ventilation rates to these buildings. The third cause is as a result of the recent concern regarding the sick building syndrome and other building related illnesses which have influenced HVAC designers to improve the indoor air quality by specifying greater quantities of fresh air supply.

As a result of these factors, the contribution of energy required for heating or cooling ventilation air to the total energy consumption for the building has increased.

There are, however, practical means of reducing the ventilating air energy requirement, some of which are briefly described below.

3.2.1. *Room temperature*

Both ventilating air and fabric energy consumption can be reduced if the set point temperature in the building is reduced during the heating season and increased during the cooling season. For the UK climate, it has been estimated that a reduction of 1 K in internal temperature from that recommended by ISO 7730 [13] will reduce the energy consumption by 6% [14]. Similar reductions have been estimated for Finland [15]. Field studies of thermal comfort have shown that up to 2.4 K reduction in indoor temperature from that specified in comfort standards can be tolerated by the building occupants without adverse affects on comfort. This is due to the fact that current standards, e.g. ISO 7730 [13] and ASHRAE Standard 55 [16], recommend indoor temperatures which are based on laboratory studies but, in real buildings, clothing habits and activity levels are known to be different from those under ideal test conditions.

3.2.2. *Ventilation system balancing*

Improper balancing of mechanical ventilation systems can result in increased fresh air rates to some zones and a reduction in others. This could not only cause discomfort due to draught in over ventilated zones and poor air quality in under ventilated zones, but could also increase the ventilation energy consumption. It is, therefore, important to check the actual delivery of fresh air to different zones of the building during commissioning and routine maintenance to ensure optimum operation of the ventilation system. Another source of energy wastage is leakage from ventilation ducts. This can also be reduced by specifying better ducts and exercising quality control during installation.

3.2.3. *Heat recovery*

In mechanically ventilated buildings, heat recovery from ventilation air is the single most important means of reducing ventilation energy consumption. Many different types of heat recovery systems are available for transferring energy from the exhaust air to the supply air or vice versa. The most commonly used systems are the regenerative type (thermal wheel), plate heat exchanger and the run-around-coil. Heat recovery of up to 70% can be achieved depending on the system used and the enthalpy or temperature difference between supply and exhaust air.

3.2.4. *Demand controlled ventilation*

Demand controlled ventilation (DCV) is a method of controlling fresh air supply to a room according to the pollution load present in the room. Although there are many pollutants that could be produced in a room due to building materials, furnishing, equipment and people's activities, it is impractical to use all these different pollutants to control the amount of fresh air supply to the room. Usually the concentration in

the room of a few types of pollutants are controlled by the DCV system. These are carbon dioxide (CO_2), TVOCs, smoke and moisture, but for most buildings the CO_2 concentration in the room is used to control the quantity of fresh air supply using CO_2 sensors which control the fresh air dampers. By controlling the fresh air supply to achieve a maximum allowable CO_2 concentration (e.g. 1000 ppm), it would be possible to reduce the ventilation rate during low or no occupancy, thus saving energy.

Although considerable amounts of research results on DCV have been accumulated, there is still a lack of experience in the installation and operation of DCV systems. Another drawback of DCV is the fact that usually only one source of pollutant (CO_2) is used for controlling fresh air rate but in normal buildings there is usually a combination of pollutants produced at different rates depending on the activities within the buildings.

3.2.5. *User control ventilation*

When designing a conventional heating and ventilation system, the energy and the air charge rate requirements are normally based respectively on a heat and pollution concentration balance over the whole space for a 'typical' day. However, there is evidence to suggest that the neutral or comfort temperature for occupants can vary substantially within the same building depending on clothing and activity of the occupants. Therefore, maintaining a whole building at the same temperature and the same fresh air supply rate can be energy wasteful. A substantial saving in energy may be achieved by individual and automatic control of the local environment by providing personalized or task-conditioning systems which can be controlled by a single user according to his or her needs. Although the capital cost of such systems is currently higher than conventional systems, this may be outweighed by the improved comfort and increased productivity of the occupants, in addition to energy saving for heating, cooling and ventilation.

4. Air flow principles

4.1. *Fluid forces*

A fluid particle in motion obeys the same laws of mechanics as a solid body, i.e. the force acting on the particle can be predicted from Newton's law of motion. Hence, the inertia force F_i acting on a moving particle is given by:

$$F_{\mathrm{i}} = m\,\partial v/\partial t$$

where m is the mass of particle (kg), v is the velocity (m s^{-1}) and t is the time (s). In addition to inertia forces, fluids in motion also experience viscous forces due to the viscosity of the fluid. The shear stress τ is given by Newton's law of viscosity, which is:

$$\tau = \mu\,\partial v/\partial y$$

where μ is the dynamic (absolute) viscosity of the fluid (Pa s^{-1}) and y is the distance normal to the flow direction. The shear force (F_s) is:

$$F_s = A\tau = \mu A\, \partial v/\partial y$$

In a moving fluid, both of the forces F_i and F_s are significant to different degrees. The ratio F_i/F_s is a non-dimensional number called the Reynolds number, viz:

$$Re = \rho v y/\mu \tag{3}$$

where ρ is the fluid density (kg m^{-3}). For small values of Re the viscous forces are dominant which restrict the movement of the fluid particles to follow the main flow direction, such a flow is called laminar flow. As Re increases, however, the inertia forces acting on the fluid particles dominate the weak shear forces and the flow is said to be turbulent. The transition between laminar to turbulent flow is identified by the value of Re corresponding to the nature of flow and geometry of the object if present in the flow. In practice, for flow in a smooth straight pipe an $Re \leqslant 2000$ usually suggests laminar flow but for a higher Re a transition range is usually defined followed by a fully turbulent flow for $Re > 4000$.

The momentum of a fluid particle is:

$$M = mv$$

If there were a change in momentum (or velocity) of the moving particle, then the force which causes this change is given by:

$$F = \partial M/\partial t = \partial (mv)/\partial t = \dot{m}\, \partial v$$

where $\dot{m}$ is the mass flow rate (kg s^{-1}). This equation shows that the force is the rate of change of momentum with respect to time.

4.2. *Continuity of flow*

In the absence of nuclear processes, matter is conserved. In fluid flow, the law of conservation of mass means that:

mass of fluid entering a control volume per unit time
= mass of fluid leaving a control volume per unit time
+change in the mass of fluid in the control volume per unit time

i.e.

$$(\partial m/\partial t)_{\text{in}} = (\partial m/\partial t)_{\text{out}} + V\, \partial \rho/\partial t$$

where V is the control volume (m^3) and ρ is the fluid density.

For incompressible flow, i.e. when changes in fluid density are small, which is the case for air flow in buildings, the flow continuity gives:

$$\dot{m} = \rho_1 v_1 A_1 = \rho_2 v_2 A_2$$

and since $\rho_1 = \rho_2$ for incompressible flow, then:

$$Q = v_1 A_1 = v_2 A_2 \tag{4}$$

where $\dot{m}$ is the mass flow rate (kg s^{-1}), Q is the volume flow rate (m^3 s^{-1}), v_1 and v_2 are the inlet and outlet velocities (m s^{-1}) and A_1 and A_2 are the inlet and outlet areas normal to the velocity direction (m^2).

4.3. *Pressure*

Pressure is the normal force exerted by a fluid per unit area. At any point in the fluid the component of pressure in any direction is constant, which is the static pressure of the fluid at that point. Since a fluid has a density, the pressure within a static column of a fluid will increase with depth due to gravity acting on the mass of fluid in the column. The variation in static pressure, p_y vertically is given by:

$$\delta p_y = -\rho g \delta y \tag{5}$$

Hence, the difference in static pressure between two horizontal planes at positions y_1 and y_2 is:

$$p_2 - p_1 = -\rho g(y_2 - y_1)$$

Hence, the static pressure at any horizontal plane (p_y) is:

$$p_y = p_o - y\,\partial p/\partial y$$

where p_o is the static pressure at a reference plane, y is the vertical distance above the reference plane.

For a constant increase in temperature with height $\partial T/\partial y$, i.e.

$$T = T_o - y\delta T$$

where T_o (K) is the temperature at a reference point, T is the temperature at a height y measured above the reference point and δT is the increase in temperature per m (K m^{-1}), there will be a corresponding decrease in pressure. Using eqn (5) and the gas law $p = \rho RT$, the vertical variation in pressure due to a uniform increase in temperature becomes:

$$\partial p/\partial y = -\rho_o g T_o/T \tag{6}$$

where ρ_o is the fluid density at a reference temperature T_o. The pressure difference between two vertical points 1 and 2 at temperatures T_1 and T_2 separated vertically by a distance y is given by:

$$p_2 - p_1 = \rho_o g T_o y[1/T_2 - 1/T_1] \tag{7}$$

The pressure in a moving fluid has a static component and a kinematic component, i.e. the total pressure p_t of a moving fluid particle is:

$$p_t = p + p_s$$

where p_s is the static pressure and p is the kinematic (velocity) pressure which is $1/2\ \rho\ v^2$. The total pressure of a moving fluid can be measured using a pitot tube with the

static pressure measured using a static pressure tap with the opening parallel to the flow direction.

4.4. *Bernoulli's equation*

The energy balance of a fluid flow without a change in temperature (isothermal) is represented by Bernoulli's equation, viz:

$$p_1 + 1/2\rho_1 v_1^2 + \rho_1 g y_1 + \Delta p_{\text{in}} = p_2 + 1/2\rho_2 v_2^2 + \rho_2 g y_2 + \Delta p_f \tag{8}$$

where

p_1 and p_2 are the static pressure at inlet and outlet, Pa
v_1 and v_2 are the velocities at inlet and outlet, m s^{-1}
ρ_1 and ρ_2 are the fluid densities at inlet and outlet, kg m^{-3}
y_1 and y_2 are the heights of inlet and outlet from a datum, m
Δp_{in} is the pressure rise due to a fan or a pump, Pa; and
Δp_{f} is the pressure loss in the system due to friction and flow separation, Pa.

5. Building air leakage and natural ventilation

5.1. *Flow characteristics of openings*

The air flow openings in buildings are of two types: adventitious openings and ventilation openings. Adventitious openings are present in every building to a different degree depending on the method of construction and installation of services. They range from gaps at wall/ceiling and wall/floor joints to openings associated with electric, water, gas services, etc. Operable building components such as doors and windows can also allow air penetration through interfaces and gaps. Ventilation openings are purposely installed to provide air supply or extract through the building, such as openable windows, air vents and stacks.

The flow through small openings, such as cracks and joints, is either laminar or transitional and that through large openings, such as ventilation openings, is usually turbulent.

For very small openings where the flow is laminar, the pressure drop is represented by the Couette flow equation, viz:

$$\Delta p = 12\mu LQ/(bh^3) \tag{9}$$

where

L is the depth of openings in flow direction, m
Q is the flow rate through the opening, m^3 s^{-1}
b is the width of opening, m
L is the height of opening, m; and
μ is the dynamic viscosity, Pa s

If the flow is transitional, i.e. neither fully laminar nor fully turbulent, the following power law equation can be used:

$$\Delta p = [Q/(kL)]^{1/n} \tag{10}$$

where

k is a flow coefficient, which is dependent on the geometry of the opening, $m^3\ s^{-1}\ m^{-1}\ Pa^{-1}$
L is the length of opening, m; and
n is a flow exponent dependent on the flow regime.

For a laminar flow $n = 1$, for a turbulent flow $n = 0.5$ and for a transitional flow n is usually between 0.6 and 0.7.

For turbulent flow through large openings, the pressure drop is given by the following equation:

$$\Delta p = 0.5[Q/C_d A)]^2 \tag{11}$$

where A is the physical area of the opening and C_d is the discharge coefficient which depends on the sharpness of the opening and the Reynolds number, *Re*. For a sharp opening $C_d \approx 0.6$, which is independent of *Re*.

5.2. *Wind and buoyancy pressures*

5.2.1. *Wind pressure*

The time-mean pressure due to wind flow on to or away from a surface is given by:

$$p_w = 0.5 C_p \rho v^2 \tag{12}$$

where

C_p = static pressure coefficient with reference to the static pressure upstream of the opening
v = time-mean wind speed at datum level (usually height of building or opening), $m\ s^{-1}$
ρ = air density, $kg\ m^{-3}$.

C_p is usually obtained from wind tunnel measurements using a scaled model of the building or using CFD. It can have a positive (e.g. windward face) or a negative (leeward face) value.

The wind speed is a temporal and a spatial varying quantity as a result of wind turbulence and the effect of natural or man made obstructions. For a given location and at a given instant the wind speed increases with height above ground where it is essentially zero. The wind speed profile is usually expressed by:

$$v/v_r = cH^a \tag{13}$$

where

v = time-mean speed at height H above the ground, $m\ s^{-1}$

Table 3
Terrain factors

Terrain	c	a
Open flat country	0.68	0.17
Country with scattered wind breaks	0.52	0.20
Urban	0.35	0.25
City	0.21	0.33

v_r = time-mean wind speed measured at a weather station normally at a height of 10 m above the ground, m s^{-1}; and
c and a are factors which depend on the terrain.

Values of c and a are given in Table 3.

Values of v_r which represent hourly mean wind speeds which are not exceeded 50% of the time can be usually obtained from a local weather station or from wind contour maps for the country.

5.2.2. *Stack pressure*

Using eqn (7), the stack pressure difference between two vertical openings separated by a vertical distance h becomes:

$$p_s = \rho_o g T_o h[1/T_e - 1/T_i] \tag{14}$$

where

T_e = external air temperature, K
T_i = internal air temperature, K.

5.3. *Flow through openings*

To estimate the amount of air flow through an opening, it is necessary to know the pressure difference across the opening and its effective flow area. The pressure at an opening can be due to wind, as well as buoyancy and it is, therefore, determined by the location of the opening in the building, as well as the internal and external environmental parameters such as wind velocity, wind turbulence and inside and outside air temperatures. The buoyancy pressure is given by eqn (14). However, the wind pressure is usually a fluctuating force which induces time-mean flow through an opening and a fluctuating (pulsating) flow through it. These two components are normally treated separately because they require different methods of calculation. The time-mean flow through an opening due to wind or buoyancy is given by either eqn (10) or (11) depending on the type of opening. Usually, eqn (10) is used for small adventitious openings and eqn (11) is used for ventilation openings. In both equations time-mean pressure differences across the opening due to buoyancy and wind is required.

Alternatively, a quadratic summation of the flow rate due to wind and buoyancy may be made using:

$$Q_t = \{Q_w^{1/n} + Q_s^{1/n}\}^n$$

where n has a value of 2/3 for cracks and 1/2 for large openings.

For a fluctuating wind pressure the resulting flow through an opening can be very complex, where inflow and outflow can occur either simultaneously or alternately depending on wind turbulence, opening geometry and internal pressure. The physics of this flow are explained in Etheridge and Sandberg [5] and the effect of a pulsating flow through windows is discussed in Section 5.4.

For ventilation (large) openings the total flow rate Q_t taking into consideration wind, buoyancy and mechanical ventilation due to a fan can be calculated using:

$$Q_t = \{Q_w^2 + Q_s^2 + Q_{mu}^2\}^{1/2} \tag{15}$$

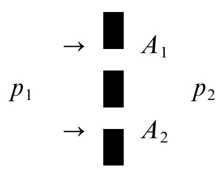
where Q_w, Q_s and Q_{mu} refer to the flow rate due to wind, stack and unbalanced mechanical ventilation (i.e. supply or extract fan).

If there is more than one opening through which air is flowing, then the effective flow area will depend on the position of the openings in the flow direction. If the openings are on one surface which is exposed to the same pressure then the effective area of the openings is given by:

$$A_{eff} = A_1 + A_2 + A_3 + \ldots \tag{16}$$

and the pressure difference across each opening is

$$\Delta p = p_1 - p_2 \tag{17}$$

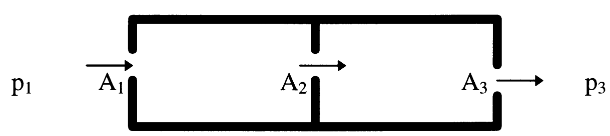


On the other hand, if the flow openings are in series, see figure below, the effective area is obtained using:

$$1/A_{eff}^2 = 1/A_1^2 + 1/A_2^2 + 1/A_3^2 + \ldots \tag{18}$$

and

$$\Delta p = p_1 - p_3 \tag{19}$$



The flow rate through multi-openings can be calculated by substituting eqns (16–19) into eqns (9), (10) or (11), depending on the flow regime.

5.4. *Single sided ventilation*

The air flow through a large single ventilation opening, such as a window, in a room which is otherwise air tight is bi-directional. The effect of buoyancy is such that cooler air enters at the lower part and warm air leaves at the upper part of the opening. The wind pressure has a mean and a fluctuating component due to turbulence. For a large opening, both the mean and the fluctuating pressure components may not be uniform over the opening. A further complication is the effect of compressibility of room air. An analytical solution of this problem is not yet available. However, air change measurements for flow through open windows carried out by de Gids and Phaff [17] on site for various wind speeds, produced the following empirical expression for the effective velocity, v_{eff}:

$$v_{\mathrm{eff}} = \{c_1 v_r^2 + c_2 H \Delta T + c_3\}^{1/2} \tag{20}$$

where

c_1 = dimensionless coefficient depending on the window opening ≈ 0.001
c_2 = buoyancy constant ≈ 0.0035
c_3 = wind turbulence constant ≈ 0.01
v_{r} = mean wind speed for the site measured by a weather station, m s^{-1}
H = height of opening, m
T = mean temperature difference between inside and outside = $T_{\mathrm{i}} - T_{\mathrm{o}}$, K

The flow rate through the opening is:

$$Q = 0.5 A v_{\mathrm{eff}} \tag{21}$$

where

A = effective area of open window, m^2.

For a single-sided ventilation BRE Digest 399 [18] recommends a window area of about 1/20 flow area and maximum room depth of 2.5 times the ceiling height, Fig. 2.

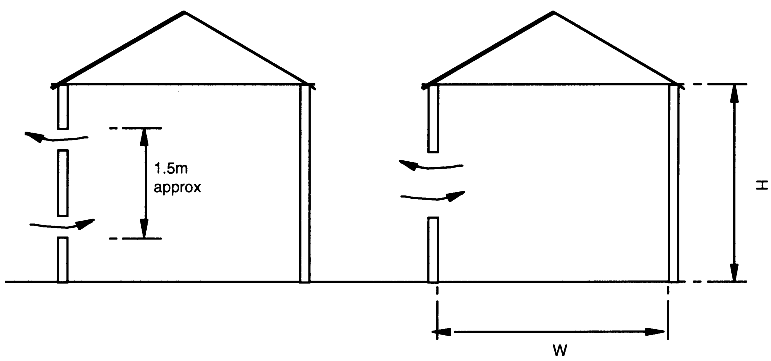


Fig. 2. Single-sided ventilation.

5.5. *Cross ventilation*

Two-sided or cross ventilation occurs when air enters the room or building from one or more openings on one side and room air leaves through one or more openings on another side of the room or building. The flow of air in this case is due to wind and buoyancy pressures. The types of openings that are used for cross ventilation can be small openings such as trickle vents and grilles, or large openings such as windows and doors. Because the air flow 'sweep' the room from one side to the opposite side, it has a deep penetration. This method is, therefore, more suitable for ventilating deep rooms. The position of openings should be such that some are placed on the windward facade of the building and others placed on the leeward facade so that a good wind pressure difference is maintained across the inflow and outflow openings. Internal partitions and other obstructions can affect or disturb the airflow pattern in the room and the air penetration depth.

The air flow rate due to cross ventilation may be estimated using eqn (11). The pressure difference across the opposite openings Δp is calculated for the combined effect of wind and buoyancy. The effective area in eqn (11) is calculated using eqn (18) and the discharge coefficient C_d depends on the type of opening. If no value for C_d is given for the opening, a value of 0.65 for a sharp-edge orifice should be used.

For cross ventilation BRE Digest 399 [18] recommends a maximum room depth of 5 times ceiling heights in a room with few obstructions, Fig. 3.

5.6. *Stack ventilation*

Buildings which require ventilation rates greater than those achievable using either single-sided or cross ventilation may be ventilated using stacks. In this case, buoyancy is the main driving force and, therefore, the height of the stack becomes significant. The stack pressure which will be determined by the difference between the internal and external temperature and the height of stack, is given by eqn (14).

Depending on the position of air inlet and outlet in the building, the wind pressure

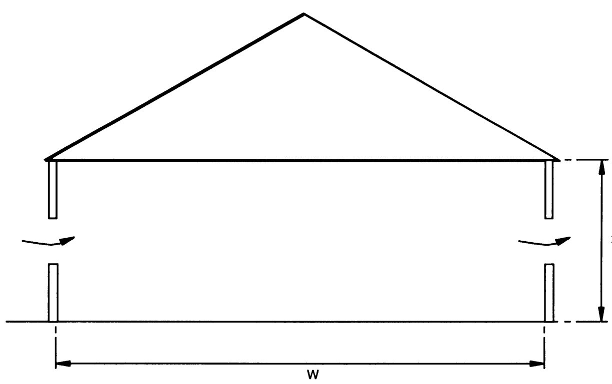


Fig. 3. Cross ventilation.

could assist the stack pressure, reduce its influence or indeed reverse the effect, i.e. by forcing the air through the outlet. Therefore, careful considerations are needed when stacks are incorporated in the building design to avoid these adverse effects occurring. This usually requires either a wind tunnel investigation of a scaled model of the building and the stack or CFD analysis of the flow around and within the building. The effect of buoyancy cannot be modeled in a wind tunnel but it can be taken into account in a CFD simulation of the air flow.

In buildings which have atria attachments, the stack is most conveniently incorporated with the atrium for two main reasons. Firstly, the solar gain in the atrium causes an elevation of the air temperature and hence there will be more effective stack flow. Secondly, the atrium will act as a buffer zone between the building and the external environment which can reduce heat losses from the building in winter.

6. Solar-induced ventilation

Natural ventilation systems are usually designed on the basis of a buoyancy-driven flow to provide a margin for variation from the expected environmental conditions. In situations where the wind assist the buoyancy flow, there should be little difficulty in providing the required air flow rate to the building. However, in the cases where the normal buoyancy pressure (resulting from the difference between the internal and external air temperatures) is not sufficient to provide the required ventilation rates then solar-induced ventilation can be a viable alternative. This method relies upon the heating of part of the building fabric by solar irradiation resulting into a greater temperature difference, hence larger air flow rates, than in conventional systems which are driven by the air temperature difference between inside and outside.

There are usually three devices which can be used for this purpose:

- Trombe wall
- solar chimney
- solar roof

These devices are governed by the same physical principles and are based on the same fluid flow and heat transfer equations. They are described here after the underlying principles of these devices are presented first.

6.1. *Sizing of solar-induced ventilation systems*

Solar-induced ventilation is buoyancy-driven by the use of a solar air collector and, therefore, all the equations derived earlier for buoyancy pressure, eqn (14), and flow rate through large openings, eqn (11), also apply here. However, the external temperature in eqn (14) is replaced by the exit temperature of the collector. In addition, there will be pressure losses through the collector as well as pressure losses at the inlet and outlet openings.

For an air collector which is equipped with a flow control damper the pressure losses are given by the expression below:

$$\Delta p = \{4fH/D_h + K_i(A/A_i) + K_d(A/A_i) + K_e(A/A_e)\}1/2\rho_m v_m^2 \quad (22)$$

where

A = cross-sectional area of ventilation channel, m^2
A_i, A_d, A_e = area of inlet, damper and exit openings respectively, m^2
K_i, K_d, K_e = pressure loss coefficients for inlet, damper and exit openings
H = height between inlet and outlet openings, m
f = friction factor for the channel
D_h = hydraulic diameter of channel, m
v_m = mean air speed through channel, m s^{-1}
ρ_m = mean air density, kg m^{-3}

The hydraulic diameter is given by:

$$D_h = 2wd/(w+d) \quad (23)$$

where

d = channel depth, m
w = channel width, m.

For a narrow channel ($w < 10d$):

$$D_h = 2d \quad (24)$$

The exit temperature, T_e of the collector is given by [19]:

$$T_e = A/B + [T_i - A/B] \exp\{-BwH/(\rho_e C_p Q)\} \quad (25)$$

where

$A = h_1 T_{w1} + h_2 T_{w2}$
$B = h_1 + h_2$

h_1 and h_2 are the surface heat transfer coefficients for the internal surfaces of the channel and T_{w1} and T_{w2} are the temperatures of the corresponding internal surfaces of the channel.

T_i = inlet air temperature of collector, °C
Q = volume air flow rate, m^3 s^{-1}
ρ_e = air density at exit, kg m^{-3}
C_p = specific heat of air, J kg^{-1} K^{-1}

The heat transfer coefficients h_1 and h_2 are usually obtained using

$$Nu = 0.1 Ra^{1/3} \quad (26)$$

where

$Nu = hH/k$ = Nusselt's number
$Ra = PrGr$ = Rayleigh number
$Pr = \mu C_p/k$ = Prandtl's number

$Gr = g\beta H^3(T_w - T_i)/\nu$ = Grashof's number
μ = dynamic viscosity of air, Pa s
k = thermal conductivity of air, W m^{-1} K^{-1}
ν = kinematic viscosity of air, m^2 s^{-1}
β = cubic expansion coefficient of air $\approx 1/T_i$, K^{-1}

Equation (26) applies to vertical and moderately inclined surfaces ($<30°$ from the vertical) for a Ra range $10^{13} > Ra > 10^9$.

Equations (22), (25), (14) and (11) are all interconnected and to estimate the air flow rate produced by the device, Q, it is necessary to solve these four equations by iteration using a computer. In eqn (11) Δp is the difference between the stack pressure given by eqn (14) and the pressure losses in the collector given by eqn (22). Further details are given in refs [19, 20].

6.2. *Trombe wall ventilator*

A Trombe wall collector consists of a wall of moderate thickness (thermal mass) with a lower and an upper opening covered externally by a pane of glass. A gap of 50–100 mm between the glass and the wall allows the heated air to rise. Trombe wall collectors have traditionally been used for space heating by allowing air from the room to enter at the bottom of the wall which is heated by the collector and then returned back to the room at high level, see Fig. 4.

The arrangement shown in Fig. 4 is for the winter situation where the Trombe wall is used to heat room air. However, by putting a high level external opening on the glazing and closing the top opening to the room this device can be used for cooling the room by drawing outdoor air from another opening into the room and the warm

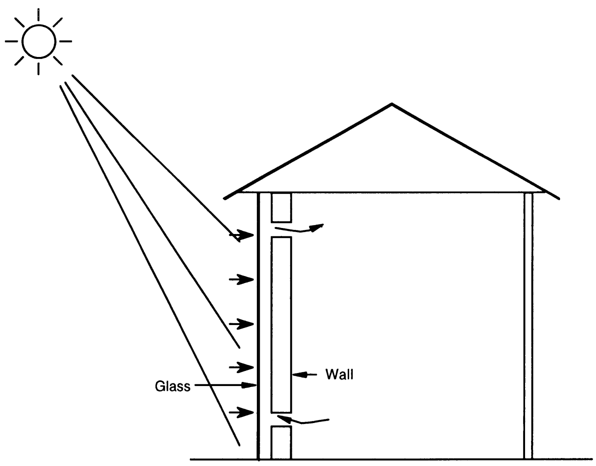


Fig. 4. Trombe wall collector for heating.

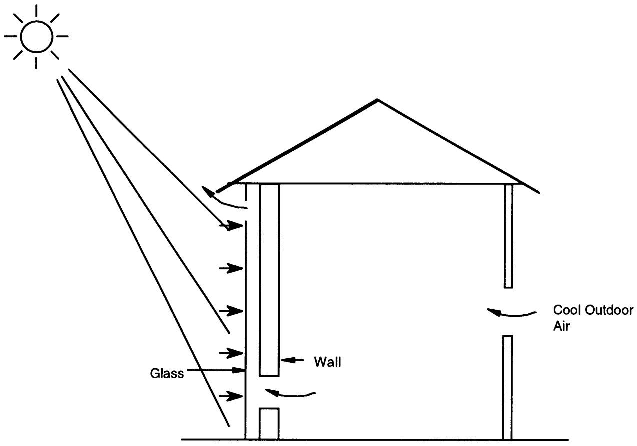


Fig. 5. Trombe wall ventilator.

air is extracted out through the Trombe wall, Fig. 5. To be effective, the wall needs to be placed in a south or south-west facing position in the northern hemisphere.

To calculate the air flow rate through the collector the method described in the previous section is applied. However, an estimate of the wall and glazing temperature will be required and these can be estimated from a knowledge of the solar gain, thermal mass of wall, emissivity of glass and wall, etc. For this purpose the reader should refer to publications on the design of Trombe walls [21, 22].

6.3. *Solar chimney*

A solar chimney attached to south/south-west facing wall is heated by solar irradiation and the heat stored in its fabric can be utilised for ventilation, Fig. 6.

The heated external surface of the chimney generates a natural convection current by drawing air from the building and extracting it at the top. Outdoor air enters the building to replace the warm,stagnant air inside.

The method described in Section 6.1. also applies here but usually only the external surface of the chimney is heated. In this case, eqn (25) may be simplified to:

$$T_e = T_w + (T_i - T_w)\exp\{-hwH/(\rho_e C_p Q)\} \tag{27}$$

where T_w = the inside wall temperature of chimney, °C.

In designing a solar chimney particular attention should be given to the depth, i.e. the gap between the chimney and the building. As the gap increases, the air flow rate increases but when the gap exceeds a certain value the flow rate starts to decrease slightly. In an experimental facility in which two surfaces of the chimney were heated the optimum gap was found to be 200 mm [23].

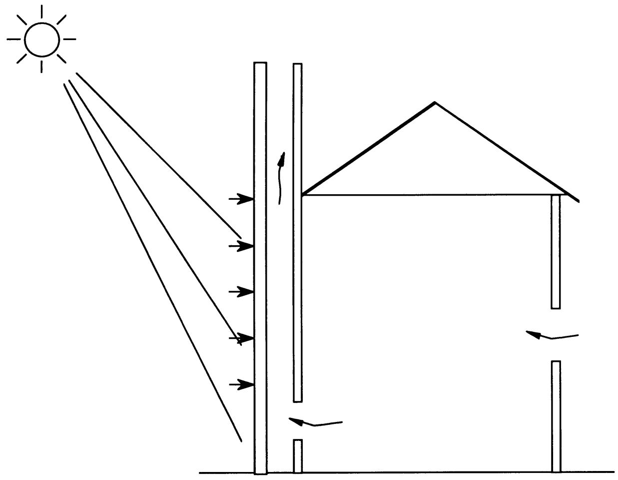


Fig. 6. Solar chimney.

6.4. *Solar roof ventilator*

In climates where the solar altitude is large, a Trombe wall or a solar chimney may not be very effective collectors of solar energy and, therefore, the ventilation rate that can be achieved with these devices may be limited. In this situation, a sloping roof collector can be more effective in collecting solar energy but because of the sloping surface the height of the collector will be small. A solar roof ventilator is shown in Fig. 7.

The advantage of a roof collector is that a large surface area is available to collect

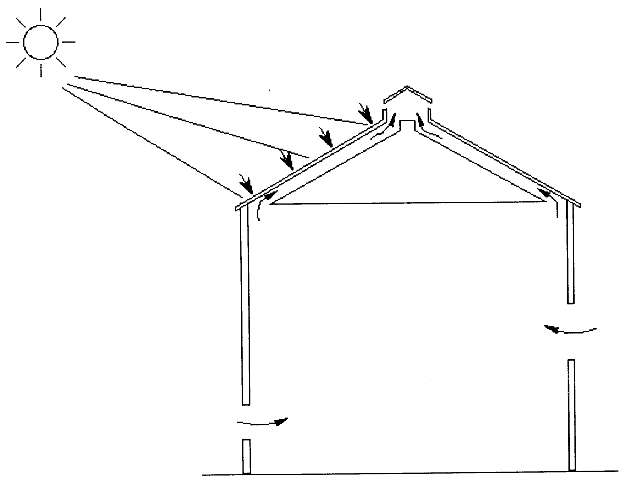


Fig. 7. Solar roof ventilator.

the solar energy and hence higher air exit temperatures can be achieved than that for a Trombe wall or a solar chimney. As a result, a roof ventilator could achieve ventilation rates close to a solar chimney or even higher depending on its design and the climate.

The estimation of the flow rate is carried out using the method given in Section 6.1. Here the height, H, is taken as the vertical distance between the inlet and outlet to the roof and not the length of the roof.

7. Mechanical ventilation

Mechanical ventilation is the provision of outdoor air or extraction of room air by the use of one or more fans. Unlike natural ventilation this form of ventilation offers the ability to control the air flows within a building according to requirements and it is essentially independent of the external weather conditions.

There are two types of mechanical ventilation systems: unbalanced systems and balanced systems. In unbalanced systems the air is either supplied to the building, or extracted from it using a fan. In a balanced system the air is supplied and extracted simultaneously using fans.

In the mechanical extract system, fans remove the air from various locations within the building through ducts and the air which is extracted is replaced by air leakage through windows, purpose-provided openings or cracks in the building envelope. This type of system is effective in removing the pollutants at their points of generation. However, as a result of the depression (negative pressure) created within the building, back flow from flues, sanitary vents, etc. could be induced. Heat recovery from the exhaust air can be achieved using a heat pump to heat water for domestic or central heating use. These systems are also suitable for extracting moisture in domestic buildings, such as from kitchens and bathrooms.

In a mechanical supply system, a positive pressure is created within the building and indoor air is forced to leak out through openings and cracks in the building fabric. This system allows the cleaning and filtration of the supply air and prevents the ingress of outdoor air through adventitious openings and is suitable in areas where the outdoor air is polluted such as in large towns. This system, however,does not allow the use of a heat recovery device because indoor air leaves the building from many locations. It is used in larger buildings where a balanced system or an air conditioning system is considered too costly to install.

Balanced systems provide both mechanical supply of outdoor air and mechanical extract of indoor air. These systems can also be used as air cleaning and heating systems by the use of air filters and pre-heaters. They are most suited to heat recovery where heat from the exhaust air is used to heat the supply air using a variety of air-to-air heat recovery devices. These systems can provide a good control of air supply and extraction to a building and they should ideally be only installed in airtight buildings. They can also be used in conjunction with a separate heating system such as a hot water radiator system. By slightly pressurising the building (i.e. a lower extract rate than the supply rate) only treated air will enter the building. However,

these systems are rather expensive and have a relatively short life cycle (typically 15–20 years) compared with the life cycle of the building. Furthermore, they require regular maintenance to ensure correct operation and good air quality in the building.

In all systems involving mechanical air supply, the indoor air quality is not only influenced by the quality of air supply but to a great extent it is also influenced by the air flow pattern in the room. The latter is determined by the type of air distribution system used for supplying the air to the ventilated space. The most widely used mechanical air distribution systems are briefly described in the coming section. However, a fuller account of these systems are given in Awbi [8] and Etheridge and Sandberg [5].

7.1. *Air jets*

Outdoor or processed air delivered to a room always enters the room as a jet. A jet of air is the flow resulting from the interaction of the fluid issuing from an opening with the surrounding fluid. This process is called entrainment of the secondary fluid (fluid surrounding the jet) by the primary fluid (the fluid issuing from the opening). If the jet continues to flow unobstructed, it is then called a free jet and if it is attached to a surface it is called a wall jet. The development of the two types of jets is different because of the influence of the boundary layer next to the surface on the jet and also because that side of the wall jet will not entrain the secondary fluid. The momentum of a jet will ideally be conserved in the flow direction, whereas the mass flow increases as a result of entrainment. In practice, the momentum usually decreases due to the dissipation of turbulence energy and also surface friction in the case of a wall jet. However, for a confined jet there could be situations where the momentum actually increases with distance [24]. The velocity across a jet is zero at the boundaries and reaches maximum at the centre of a free jet, or close to the surface for a wall jet. Because the mass flow increases, the maximum velocity of the jet decreases with distance from the outlet.

7.1.1. *Free jet*

For a free jet, four regions may be identified where the flow has a distinct characteristic (see Fig. 8):

(i) In Zone I the maximum jet velocity, U_m is the same as the velocity at the outlet, U_o. This extends to about 6 outlet diameters.

(ii) In Zone II the maximum velocity here is given by:

$$U_m \propto 1/x^n$$

where n is an index whose value is in the range 0.33–1.0 depending on the aspect ratio of the opening.

(iii) Zone III represents the fully developed flow zone which extends up to about 100 outlet diameters depending on the shape of the outlet opening. Here, the maximum velocity decays inversely with distance, i.e. $U_m \propto 1/x$.

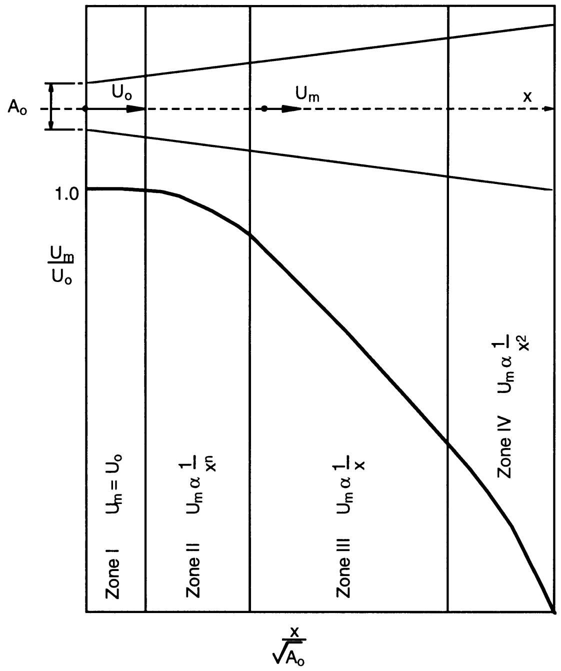


Fig. 8. Zones of air jet.

(iv) Zone IV is called the terminal zone where the velocity decays very rapidly. The maximum velocity decays with the square of the distance, i.e.

$$U_m \propto \frac{1}{x^2}$$

For a free circular jet, Zone II is small but Zone III is the most extensive zone where the maximum velocity may be represented by:

$$U_m/U_o = K_v/(x/d_o) \tag{28}$$

where K_v is called the throw constant which is in the range 5.8–7.3.

For a plane (two dimensional) jet, i.e. a jet issuing from a very long rectangular slot, Zone II is significant in which the jet velocity is given by:

$$U_m/U_o = K_v/\sqrt{(x/h)}$$

where h is the height of the opening and K_v is about 2.5.

For jets issuing from rectangular openings, the extent of Zones II and III depends on the aspect ratio of the opening. However, after about 50 $\sqrt{A_o}$, where A_o is the area of the opening, the maximum velocity decay will be given by eqn (28) for a circular jet by replacing d_o by $\sqrt{A_o}$.

7.1.2. *Wall jet*

A wall jet is most common in mixing ventilation systems (see Section 7.5) in which case a jet is directed over the ceiling. Depending on the aspect ratio of the opening, the resulting jet is either a plane wall jet from a very large aspect ratio (<40) opening or a three-dimensional wall jet from a finite aspect ratio opening. For a plane wall jet, there are two main zones, in addition to the terminal zone. The length of the first zone is about 7 h where h is the height of the slot opening. The maximum velocity in the second zone, which is the most extensive zone for a plane wall jet, is given by:

$$U_m/U_o = K_v/\sqrt{(x/h)}$$

where K_v is about 3.5.

For a three-dimensional wall jet there are three main zones, not including the terminal zone. The extent of the second zone depends on the aspect ratio of the opening, which can be up to 30 $\sqrt{A_o}$ from the opening. The velocity decay in the third zone is expressed by:

$$U_m/U_o = 14/x^{1.15}$$

In the previous discussion the jets are assumed to be isothermal (i.e. jet and room temperature are the same). If the jet is non-isothermal (i.e. hotter or cooler than room air) the development of the jet will be influenced by the Archimedes number, Ar viz:

$$Ar = g\beta d_o \Delta T/U_o^2 \tag{30}$$

The treatment of non-isothermal jets is given in [5, 8].

7.2. *Air terminal devices*

Ventilation openings are usually fitted with a device for controlling the jet and sometimes for aesthetic purposes too. Such a device is usually referred to as an air terminal device (ATD). There are many types of ATDs in use but ISO 5219 [25] classifies these according to the geometry of their openings, thus:

Class 1: Devices from which the jet is essentially three dimensional, e.g. nozzles, grilles and registers.
Class 2: Devices from which the jet flows radially along a surface, e.g. ceiling diffuser.
Class 3: Devices from which the jet is essentially two dimensional, e.g. linear grilles, slots and linear diffusers.
Class 4: Devices for generating buoyant flows, e.g. low velocity air terminals.

The flow within modern ATDs can be very complex and the jets which they produce may not behave in the same way as those jets produced by simple openings which were described earlier. The selection of ATDs should be based on the data available for each particular ATD (nomograms) normally supplied by the manufacturer. ATDs are usually tested in accordance with the procedures laid out in ISO 5219.

7.3. *Fans*

A fan is a rotary, bladed, machine producing a continuous flow of air by the aerodynamic action of the blades on the air. The performance of a fan is described by fan characteristic graphs supplied by the manufacturer. Typical fan curves would be similar to those shown in Fig. 9. To select a fan for a particular application it is necessary to know beforehand the system characteristics, i.e. the pressure losses of the system for a given flow rate. These two quantities can then be plotted on the fan characteristic graph to determine the desired duty point A, see Fig. 9.

In general, the fan curve will not pass through A and a system curve can be plotted using:

$$p/p_A = (Q/Q_A)^2$$

The point of intersection B gives the operating point, i.e. the flow rate which the fan delivers to the system. If the two points A and B are far apart, then a change in the system pressure losses will be required or a change in fan speed or a different fan is selected. Nowadays, with the availability of speed controllers, there is more flexibility in fan selection. For each speed, a different fan pressure curve is obtained. However, to operate the fan at optimum efficiency, Point B must correspond to the maximum point on the efficiency curve which is M in Fig. 9.

There are many different types of fans which suit a variety of applications in ventilation [26].

7.4. *Localized ventilation systems*

These systems supply the conditioned air directly where it is required, i.e. to the occupants. The air supply terminals are placed in the vicinity of each occupant, such as on a desk, a seat, etc. In some such systems the user has full control over his/her

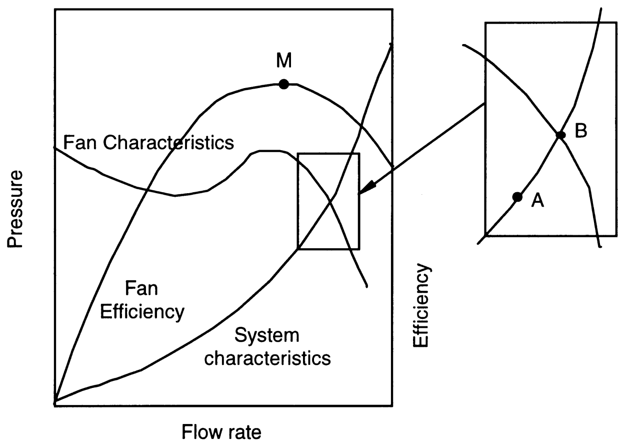


Fig. 9. Fan characteristic curves.

local environment by the ability for controlling the air flow rate, temperature, flow direction etc. Air is extracted from the occupied zone either locally or centrally. Such systems can be more energy efficient and more responsive to the needs of the individual occupant but are more costly to install and maintain. They are used in offices, theatres, hospital operating rooms and certain industrial buildings.

The air supply rates used in localized ventilation systems depend on the application. In office rooms for example the rate is determined by the fresh air requirement for the occupants and this is well documented in ventilation standards, e.g. ASHRAE Standard 62-1989R [6]. Because the fresh air is supplied directly to the occupants, generally these systems require lower air supply rates than other ventilation systems.

7.5. *Mixing (dilution) ventilation*

A mixing or dilution ventilation system aims to mix the indoor air pollutants with the supply air to achieve a uniform concentration of pollutants. This requires the supply of an air jet in addition to other forms of air movement such as plumes, convection currents, etc. produced by heat sources and room surfaces. Air is usually extracted from the room at high levels.

In most mixing systems, a wall jet is supplied over the ceiling or from a window sill opening to provide a vortex motion in the room such that high velocity air in the jet is kept within regions close to the ceiling and walls, whilst at floor level and in the centre of the room, the air velocity is sufficiently low (e.g. <0.25 m s^{-1}).

Mixing ventilation has been in use for a long time and there is a wealth of information on the design of these systems, see for example Awbi [8]. Unlike displacement ventilation (see below) mixing ventilation can be used for heating and cooling as well as providing fresh air. However, because the aim here is to provide uniform mixing of the supply air with room air, the heat emitted from internal sources such as people and equipment is fully taken into consideration when the air flow of the system is determined. The same principle also applies to the internally produced pollution. The ventilation effectiveness of mixing ventilation system (ε_v) is usually <1.0.

7.6. *Displacement ventilation*

Unlike in mixing ventilation, where the supply air is mixed with room air to dilute the pollutants, displacement ventilation tends to displace the pollution and heat in one direction, hence giving a ventilation effectiveness (ε_v) >1.0. The direction of air flow can be from the ceiling down, from the floor up, from a wall to a wall, or from a wall to the floor then up to the ceiling. Because the buoyancy effect from occupants and other heat sources causes most pollutants to rise, the ceiling supply method is less common. What is becoming more popular recently is the use of low velocity units for supplying air from a wall terminal over the floor and allowing the air to rise as it warms up by internal sources and the air is extracted from the ceiling. Such a system is shown in Fig. 10.

This type of ventilation is energy efficient because the air in the room is allowed to stratify (i.e. the air temperature increases with height) which produces the desired

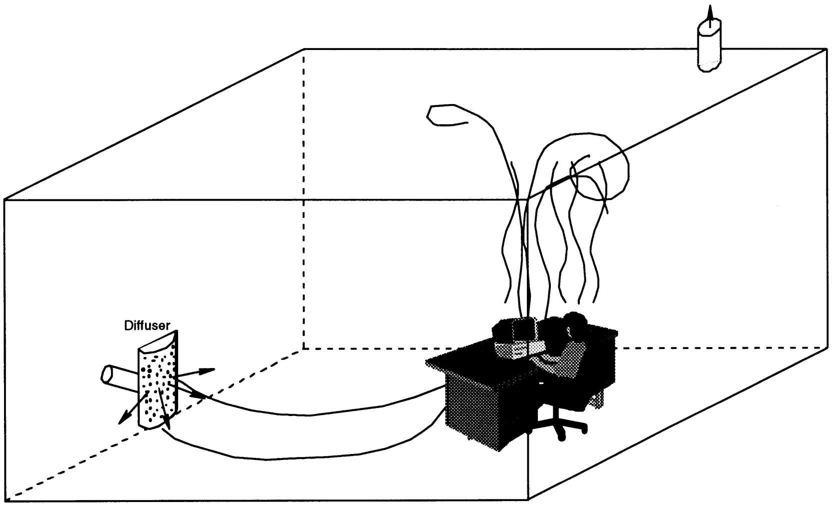


Fig. 10. A typical office room displacement system.

temperature in the occupied zone but the extract air temperature is higher. However, in a normal mixing system the extract air temperature is almost the same as the room temperature because of the mixing effect. In practice, this means supplying fresh air at low velocity (typically <0.5 m s^{-1}) near the floor directly into the occupied zone with a temperature only a few degrees (up to 5 K) below the room temperature. However, because the supply air temperature is not normally allowed to be lower than about 18°C (for comfort purposes), the cooling capacity of such a system is limited. In comparison, in a mixing system the supply temperature can be as low as 10°C which will have a higher cooling capacity than the air supplied in a displacement system. In order to overcome this limitation, some displacement systems are supplemented by chilled beams or chilled ceiling devices. A chilled beam is essentially a finned pipe carrying cold water hanging from the ceiling, whereas a chilled ceiling consists of a panel attached to a serpentine pipe containing cold water.

A major drawback with this type of displacement ventilation is that it is not suitable for heating and for this purpose a separate system is needed. Guidelines for the design of displacement systems are given in BSRIA TN 2/10 [27].

References

[1] Billington NS, Roberts BM. Building Services Engineering. Pergamon Press, 1982.

[2] Fanger PO et al. Air pollution sources in offices and assembly halls by the olf unit. Energy and Building 1988;12:7–19.

[3] Janssen JE. The V in ASHRAE: An historical perspective: ASHRAE Journal 1994;36(8):126–32.

[4] Sykes, JM. Sick building syndrome. Building Serv. Eng. Res. Technol. 1989;10(1):1–11.

[5] Etheridge D, Sandberg M. Building Ventilation: Theory and Measurement. Wiley, 1996.

[6] ASHRAE Standard 62-1989 R. Ventilation for Acceptable Indoor Air Quality. Atlanta: American Society of Heating, Refrigeration and Air-Conditioning Engineers, 1966.
[7] HSE Guidelines EH40/95. Occupational Exposure Limits. London: Health and Safety Executive, HMSO, 1995.
[8] Awbi HB. Ventilation of Buildings. London: Spon, 1991.
[9] Yaglou CP et al. Ventilation requirements. Trans. ASHVE 1936;42:133–62..
[10] BS5925 Ventilation Principles and Designing for Natural Ventilation. London: British Standards Institution, 1991.
[11] Molhave L. Volatile organic compounds, indoor air, quality and health. Proceedings of Fifth International Conference on Indoor Air Quality and Climate, Indoor Air, 1990. Vol. 5, p. 15–34.
[12] CEN Standard pr ENV 1752. Design Criteria for the Indoor Environment. Brussels: European Committee for Standardization, 1996.
[13] ISO 7730. Moderate Thermal Environments—Determination of the PMV and PPD Indices and Specification of the Conditions for Thermal Comfort. Geneva: International Organization for Standardization, 1994.
[14] Croome DJ, Gan G, Swaid H, Awbi HB. Energy implications of thermal comfort standards. Proceedings of Building Design Technology and Wellbeing in Temperate Climates, Brussels, 1993.
[15] Seppanen O. Good energy economy and indoor climate—Conflicting requirements? Proceedings of European Conference on Energy Performance and Indoor Climate in Buildings, Lyon, France, 1994.
[16] ANSI/ASHRAE Standard 55—1992. Thermal Environment Conditions for Human Occupancy. Atlanta: American Society of Heating, Refrigeration and Air-Conditioning Engineers, 1993.
[17] de Gids W, Phaff H. Ventilation rates and energy consumption due to open windows—A brief overview of research in the Netherlands. Air Infiltration Rev. 1982;4(1):4–5.
[18] BRE Digest 399. Natural Ventilation in Non-Domestic Buildings. Garston (UK): Building Research Establishment, 1994.
[19] Awbi HB, Gan G. Simulation of solar-induced ventilation. Proceedings of Second World Renewable Energy Congress, 1992;4:2016–30.
[20] Awbi HB. Design considerations for naturally ventilated buildings. Renewable Energy 1994;5(2):1081–90.
[21] Trombe F. et al. Concrete walls to collect and hold heat. Solar Age 1977;2:13–35.
[22] Akbarzadeh, et al. Thermocirculation characteristics of Trombe wall passive test cell. Solar Energy 1982;28:461–8.
[23] Bouchair, A. et al. Moving air, using stored solar energy. Proceedings of Thirteenth Passive Solar Conference, Cambridge, MA, 1988.
[24] Karimipanah T. Turbulent Jets in Confined Spaces. Ph.D. thesis. Royal Institute of Technology, Gavle, Sweden, 1996.
[25] ISO 5219. Air Distribution and Air Diffusion—Laboratory Aerodynamic Testing and Rating of Air Terminal Devices. Geneva: International Organization for Standardization, 1984.
[26] Daly BB. Woods Practical Guide to Fan Engineering. Colchester: Woods, 1990.
[27] BSRIA TM 2/90. Displacement Ventilation. Bracknell (UK): Building Services Research and Information Association, 1990.